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Applications of the Low-Pressure Steam Turbine

Mechanical Engineering

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APPLICATIONS OF THE LOW-PRESSURE STEAM TURBINE

BY

MATI LAL DUTT

THESIS

FOR THE

DEGREE OF BACHELOR OF SCIENCE

IN

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ENTITLED Applications of the Low-Pressure Steam Turbine

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE

DEGREE OF B.S. in Mechanical Engineering

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Instructor in Charge.

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Acting HEAD OF DEPARTMENT OF Mechanical Engineering

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THE LOW-PRESSURE STEAM TURBINE

I History and Development.

In considering the history and development of the low-pressure turbine, we naturally come to the steam turbine in general. The year 120 B.C saw the beginning of steam turbine practice, when Hero of Alexandria invented the Aeolpile. This engine was a reaction type of steam turbine, and although it was very crude, it utilised the reaction of flowing steam just as, for instance, Parson's turbine does at the present day. Since this time, others have attempted to utilise the energy of the steam in a similar manner, but it was not until 1884 that Mr. Parsons succeeded in building an axial reaction wheel of 10 H.P. which made 18000 revolutions, and which was a practical success, although it still needed a good many improvements before it became the machine that it is to-day. The real history of the steam turbine, therefore, dates from the year 1884. Another important date in this history is the year 1889 when de Laval brought out his turbine, in almost the same form as it is being made to-day, These two steam turbines are the foremost representatives of the two most extreme types of this

new motor. After gradual changes and continual improvements the different types of turbines of to-day have been evolved.

The history of the low-pressure turbine is a very brief one. A patent was taken out in 1894 for the combination of the Parsons' Turbine and the reciprocating engine with the object, as stated by Mr Parsons, of increasing "the power obtainable by the expansion of steam beyond the limits possible with reciprocating engines." The earliest example of a low-pressure turbine taking exhaust steam was in the torpedo boat Turbinia, but the exhaust came from a high pressure turbine and not from a steam engine. The actual combination of a reciprocating engine and a steam turbine was not put to a practical test until 1902, when two inventors tried it on radically different lines. Mr. Parsons installed it in the destroyer "Velox", where the reciprocating engines were fitted to the same shafts as the low-pressure turbines for driving the center screws. The wing screws were coupled to high pressure turbines. For cruising speed, the steam passed successively through the reciprocating engine, the high pressure turbine and the low-pressure turbine; and the economic results

were good. For high speed the engines were uncoupled and the steam passed from the boiler direct to the high and low-pressure turbines successively. About the same time Professor Rateau made the first commercial use of the combination for stationary purposes, at the Bruay mine. After this the history is a period of growth and expansion.

II Thermodynamics

As pressures were increased in steam practice the pressure difference came to be regarded as the vital point and the temperature difference was generally overlooked. This was influenced to a certain extent by the hyperbolic law of the expansion of steam, whereby low-pressure cylinders if designed in proportion to volume at each pressure would be simply impracticable and the greater percent of the work gained would be lost in overcoming the friction of heavy moving parts of the engine. The result is seen in hundreds of engines working non-condensing and exhausting steam into the atmosphere when only one-half of the total available work of the steam is realised.

The work which can be obtained from steam is entirely dependent on the amount of heat given up during the expansion. An engine gives a good efficiency in the upper part of the temperature and pressure range; in turbines, the friction of blades is high since it varies with ρ^3 where ρ = density of steam. But in the lower stages of pressure, owing to the large volume of low-pressure cylinders and the consequent cylinder condensation due to temperature changes, the efficiency of engine

is seriously reduced as the expansion is carried further down the scale. In a turbine advantage can be taken of the expansion to a point approaching the theoretical limit, each part remaining at about the same temperature continuously. Additional gain is seen in a turbine at low pressure from the fact that friction varies with v^3, and the rarer the medium the smaller the friction losses.

As an illustration of these facts, the temperature - entropy diagram may be used to advantage. Taking an ideal cycle on the Temperature - entropy plane we find, that in engines the steam at B cannot expand adiabatically to the final pressure p_2 because of very large specific volume of saturated steam. The exhaust opens and releases the steam at a pressure higher than p_2. The actual process may be represented by a constant volume line EF. This cutting of the slice CEF from the available work is justified by the smaller cylinder volume corresponding to the point F on the constant volume line EF. The point C on the constant volume line through C represents a much larger

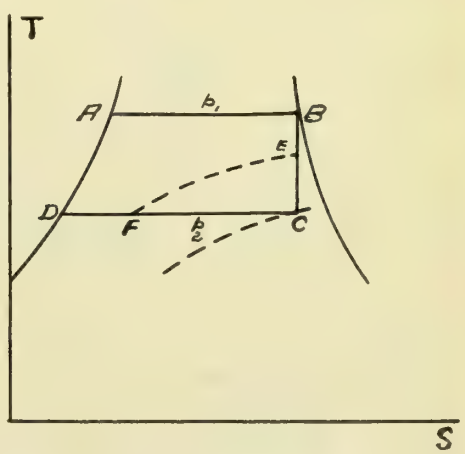


Fig. 1

volume than the volume at F. But in a turbine the steam at B could be expanded to C and the lost work ECF realised.

Again if p_1 is raised to a higher pressure p_1' the area ABB'A' is a pure gain in case of engines and consequent increased efficiency, but in turbines there is considerable loss due to friction because of the higher density of the steam.

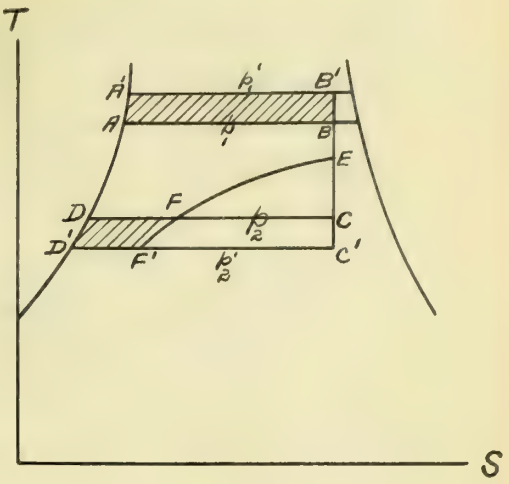


Fig. 2

If the condenser pressure p_2 is however lowered to p_2' the loss-area ECF increases to the area EC'F' and the gain is the comparatively very small area DFF'D'; but in the turbine, the total energy made available by lowering the condenser pressure is utilised. This additional energy is represented by the area DCC'D'

This discussion shows conclusively that in steam engine practice, "the limitations imposed by the cylinder volume prevents much improvement in efficiency by lowering the back pressure; whereas at upper pressures, the engine gains in efficiency to the fullest extent." In turbine practice the lower the back pressure, the greater is the efficiency.

Here in lies the importance of the low-pressure turbine.

Taking the subject quantitatively, a temperature-entropy diagram is drawn to scale in Fig 3. The several areas represent to scale the available energy per pound of dry steam The works theoretically in the several cases are:

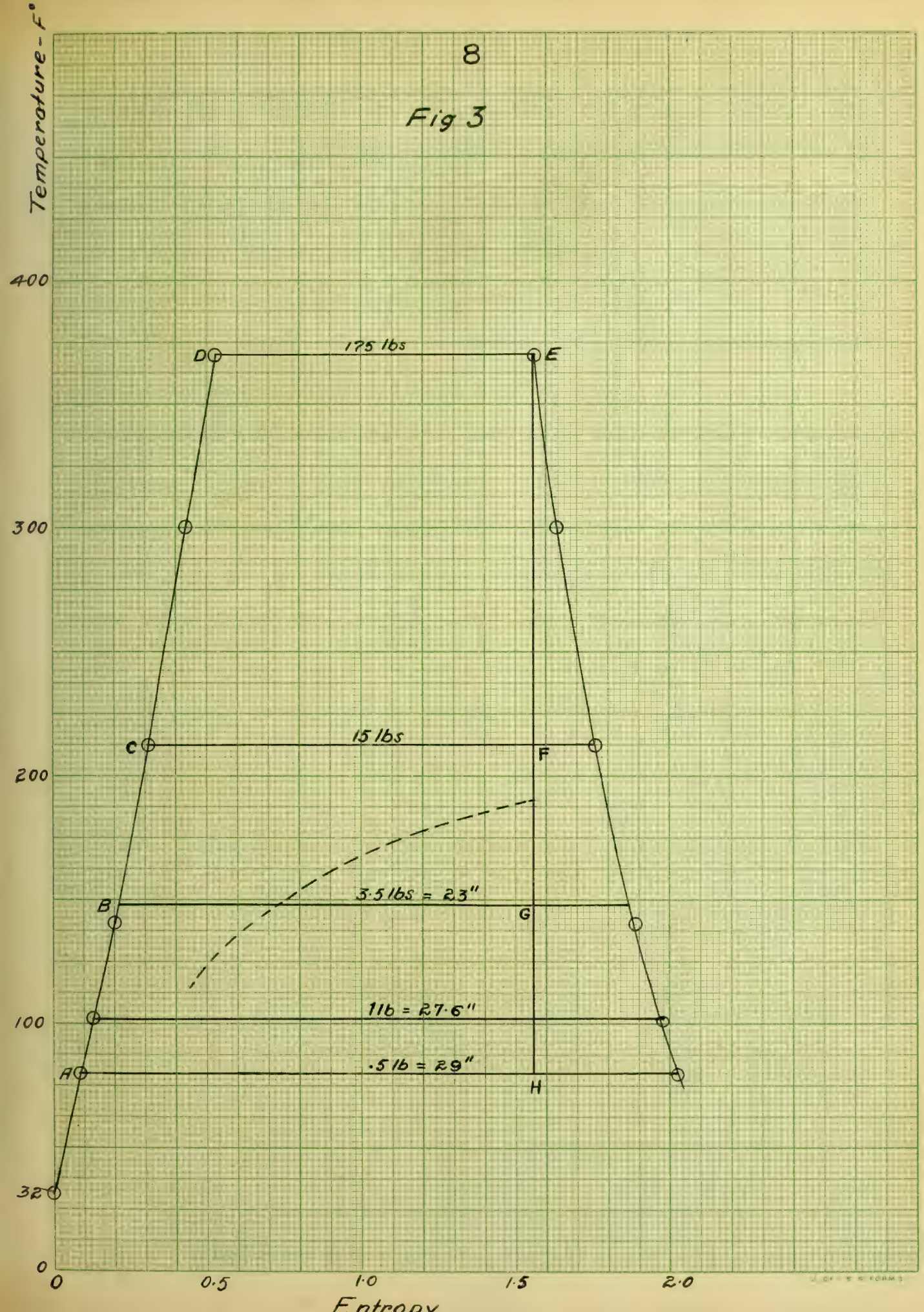
1. Area CDEF = 138437 foot-lbs, representing the case of a non-condensing engine taking steam at 175 pounds per square inch abs. and exhausting it at 15 pounds per square inch.

2. Area BDEG = 204300 foot-lbs, representing the case of an ordinary condensing engine taking steam at 175 pounds per square inch absolute and exhausting into a vacuum of 23 inches of Mercury.

3. Area ACFH = 139663 foot-lbs, representing the energy thrown away by a non-condensing steam engine in case a vacuum of 29 inches were available. In case this exhaust steam were turned into a low-pressure turbine after the moisture has been drained of it, the area should be equal to 138000 foot-lbs.

The diagram shows at a glance that there is about the same available energy between almospheric pressure and the best obtainable vacuum as there is between 175 pounds pressure and atmospheric pressure. The ordinary steam engine and condensing

Fig 3


Temperature - F°
400
300
200
100
32
0
0
0.5
1.0
1.5
2.0
Entropy
D
E
175 lbs
C
F
15 lbs
B
G
3.5 lbs = 23"
1 lb = 27.6"
.5 lb = 29"
A
H

apparatus cannot in the nature of things take advantage of this, and the great economy exhibited by the steam turbine is wholly due to the fact that it can. The three figures, 138437 for atmospheric exhaust, 204300 for condensing steam engines, and (138437+138000)= 276437 for the steam turbine, or the combined steam engine and low-pressure turbine exhausting into a first rate vacuum, tell the whole story. The steam engine and the turbine are natural companions — one for the high and the other for the low pressures — although it is extremely doubtful if the two should be designed and built for combined operation. The place of the low-pressure turbine is that of a gleaner in a very wasteful field.[1]

Classes of Turbine: their construction and principles of operation.

The principles which distinguish the different kinds of turbine are well known. However, a short discussion of the distinctive features will be given in order that the different types may be easily differentiated.

In common with all steam engines, steam turbines transform into mechanical energy the energy given out by steam during its expansion from initial pressure to final pressure. In reciprocating engines this transformation of energy is effected by means of variation of pressure, where as in turbines, there is to a certain extent a double transformation: (1) of heat energy into kinetic energy of a moving jet or stream, (2) of the kinetic energy of the jet into effective work. The employment of velocity characterises the action or impulse type, where as the use of pressure and velocity characterises the pressure or reaction turbine.

Fig. 4 shows the vane and nozzle arrangement of a turbine of the impulse type. B is the diverging nozzle, CC are the rotating blades moving in the

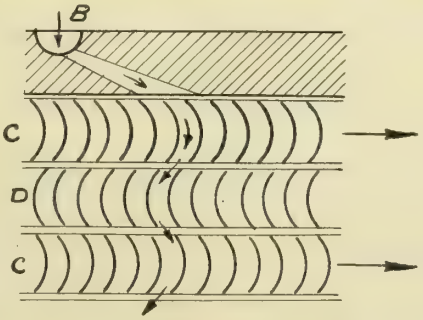


Fig 4 – Vane and Nozzle arrangement of impulse turbines.

direction of the large arrows and D are the stationary blades or reversing guides. The steam directly enters the nozzle B at boiler pressure and is completely expanded in the nozzle down to the back pressure of the exhaust, all its available pressure energy being thus converted into velocity.

The principle of operation of this type of turbine is shown by the small arrows, indicating the flow of steam. The velocity-diagram may be used to advantage in this exposition. In Fig. 5, let w_1 = velocity of steam entering from the nozzle at an angle α with the plane of the wheel Combining w_1 with C, the velocity of the blade the velocity r_1 of the jet relative to the blade is obtained. The angle β between the direction of r_1 and the plane of the wheel determines the angle of the blade at entrance. The blades being symmetrical the exit relative velocity r_2 makes the same angle β with the plane of the wheel, and in frictionless case $r_2 = r_1$. Combining r_2 and C, the resultant absolute exit velocity w_2 is obtained. The same process repeats itself in succeeding sets of vanes. The kinetic energy of the jet with velocity w_1 is $\frac{m}{2g} w_1^2$ per m pounds of

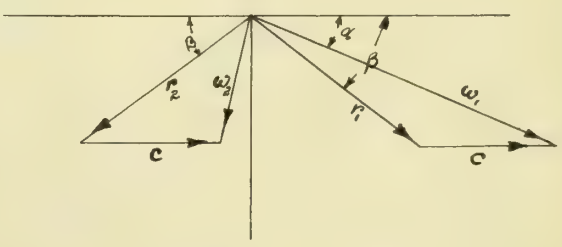


Fig 5

steam, at exit the energy = $\frac{m}{2g} \omega_2^2$. Therefore, theoretically the work absorved by the wheels, $W = \frac{m}{2g}(\omega_1^2 - \omega_2^2)$

Fig. 6 shows the arrangement of the stationary and moving buckets of the turbine of the reaction type. The stationary buckets or guides BB correspond to nozzles in the impulse type of turbine, and CC are the moving buckets similar in form to the stationary guides. The steam enters the first set of stationary buckets B at boiler pressure. Owing to the nozzle-like form of these blades, the steam here undergoes a partial drop in pressure, with a corresponding expansion and increase in velocity. The drop in pressure per stage in this type of turbine rarely exceed 10 or 12 pounds, and the velocity of the steam is not nearly as high as in the case of the greater expansion that occurs in the nozzle of the impulse turbine. The steam issues at moderate velocity, and but slightly reduced pressure, from the guides B and impinges upon the first set of moving buckets C imparting to them by impulse that portion of its pressure energy that has been converted into velocity by its passage through the guides B. Owing to the shape of the moving buckets

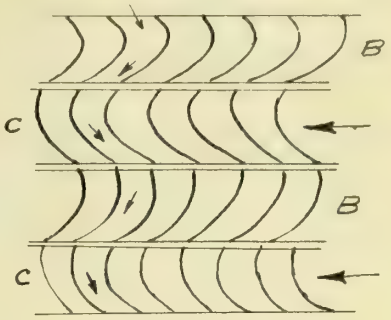


Fig 6 - Stationary and moving vanes in Reaction turbine

which have the same nozzle-like form as the guides, the steam undergoes a still further pressure drop, with a corresponding expansion and increase in velocity in its passage through the vanes. As a result of this increase in velocity, the issuing steam imparts to the buckets by reaction a portion of the pressure energy that has been converted into velocity. The same process is repeated in the second set of stationary and moving vanes, and so on, throughout as many stages as are required to abstract all the available energy from the steam.

Referring to the flow of steam through the blades and the velocity diagram, Figs. (7) and (8), let w_1 be the absolute velocity of the steam entering the fixed vanes f. The steam passing through nozzle-like vanes acquires a larger velocity w_2 due to drop in pressure. Combining w_2 with the peripheral velocity c we get r_1, the velocity of entrance relative to the moving blades. Now the pressure drops through the moving blades also; hence as a result the velocity of exit r_2 is greater than r_1. Combining r_2 with C, the result is

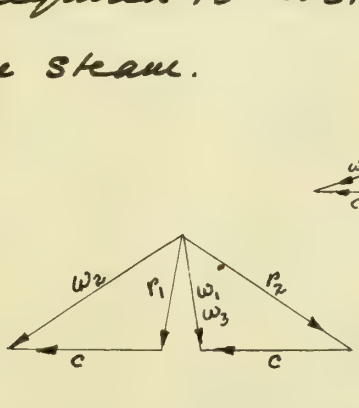


Fig. 7

Fig 8

w, the absolute velocity of entrance into next row of blades. The work done in any single stage is obtained from the velocity diagram in the usual manner as shown before.

Turbines are divided into these two principal classes:

1. Action or Impulse type — { de Laval, Curtis, Rateau, Zoelly }

2. Reaction or pressure type - { Parsons, Hamilton Holzworth }

The present de Laval turbine consists of a single bucket wheel and one set of diverging nozzles, Fig 9. The steam entering the nozzles at boiler pressure is expanded in the nozzle to the back pressure, all its available energy being converted into velocity. This velocity of the steam is utilised in driving the buckets by impulse. Low-pressure turbines of this type are not constructed.

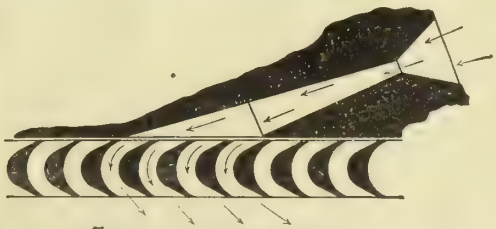


Fig 9 De Laval (Sweden)

Power

The Curtis turbine consists of one or more pressure stages and each pressure stage has sets of moving and fixed blades. Fig 10 shows the arrangement

of blades of a single stage turbine. Fig 11 shows the arrangement of a two stage turbine. The velocity of a jet is reduced by each reversal in the moving blades

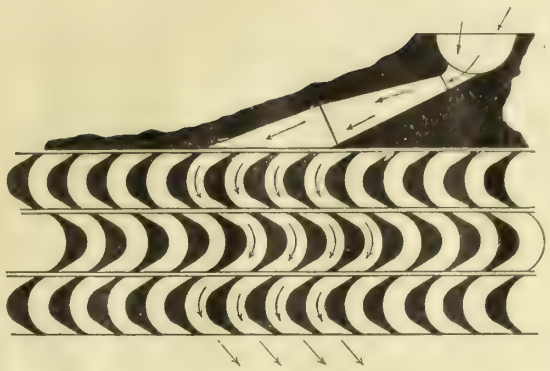


Fig 10 Curtis (United States) POWER

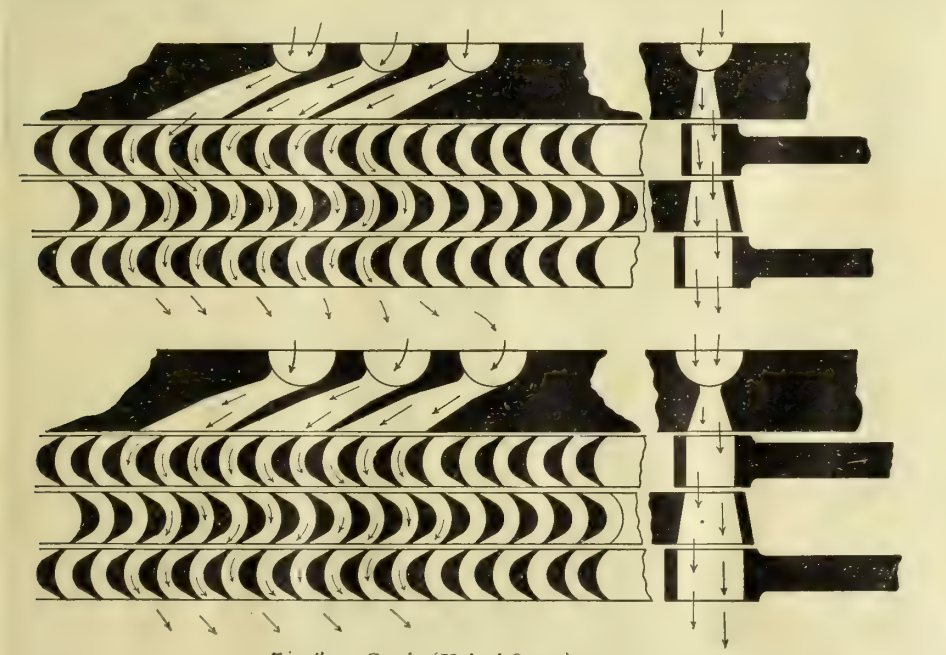


Fig. 11 Curtis (United States)
Multistage Turbines - Impulse Type POWER

and this process is carried out as many times as may be necessary to utilise the initial velocity of the jet. Thus the steam in each stage is alternately accelerated and retarded in the blades.

In the Rateau turbine, the principal of pressure compounding is employed. The turbine consists essentially of several deLaval turbines in series, running

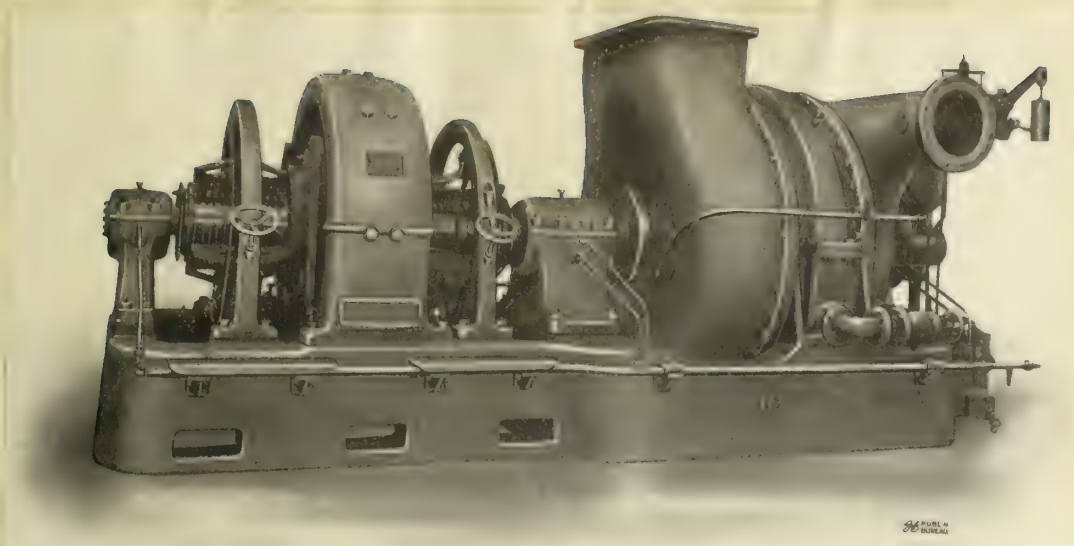


Fig 12 LOW PRESSURE CURTIS STEAM TURBINE DIRECT CONNECTED TO 500 KW. GENERATOR (TYPE CC 4-500-1500)

in separate chambers. Fig 14 shows the arrangement of blades and guides. The construction of the Zoelly type is very similar to that of the Rateau type, and all the kinetic energy developed in each fixed guide is utilised in a single revolving wheel.

In the Parsons turbine, guide vanes are employed to direct the steam against the vanes of the running wheel. It is not attempted in these guide vanes to transform the energy of the steam completely into kinetic energy, and on emerging from the guide vanes, the energy of the steam is only partly kinetic. That portion which is kinetic is more or less completely imparted to the vanes of the moving wheel. A further part of the energy of the steam emerging from the

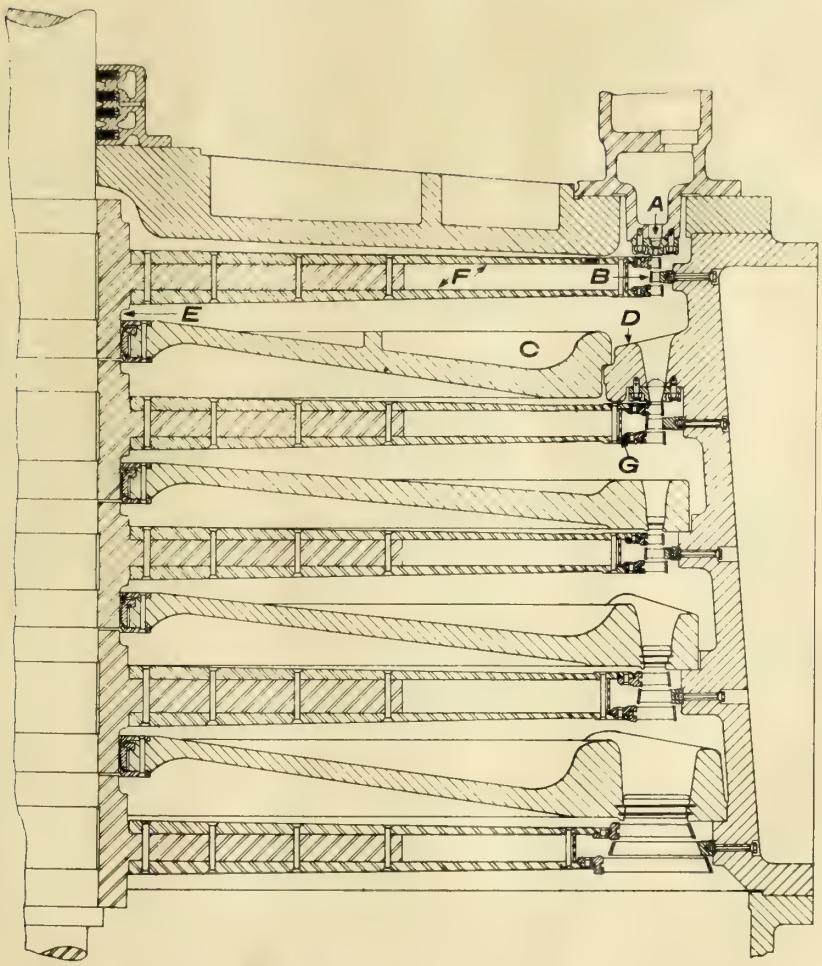


Fig. 13 Sectional View of Turbine Casing

G.E. Co.

Fig 14 Wilkinson (United States)
Rateau (France)
Zoelly (Switzerland)
Multistage Turbines - Impulse Type

Power

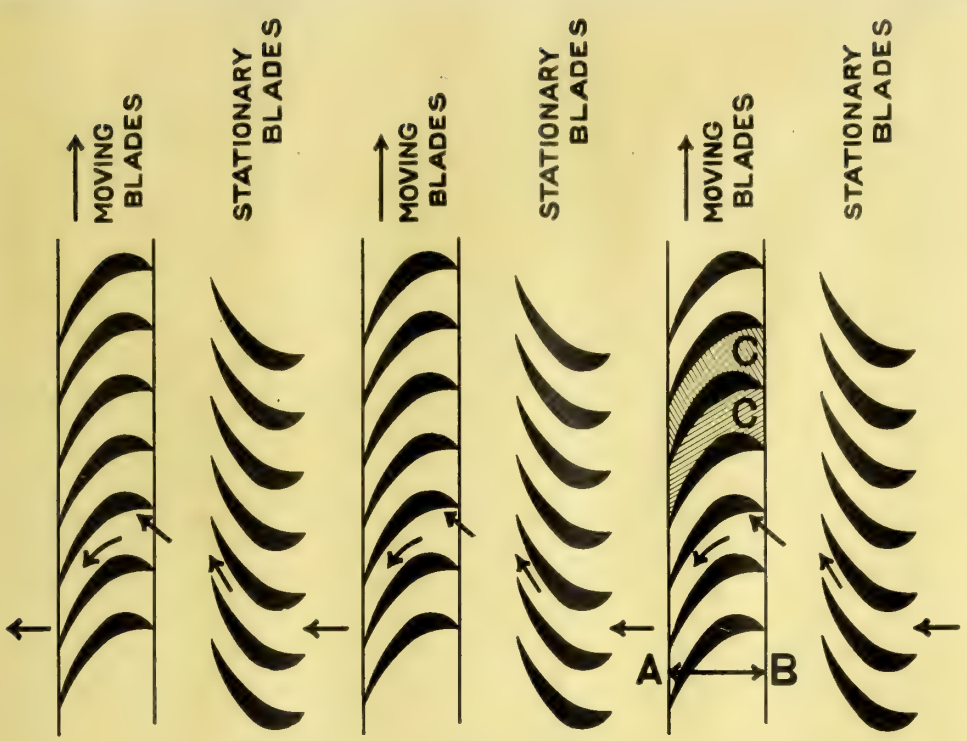


Fig. 15
Arrangement of Blading in Parsons Type Turbine.

Allis-Chalmers

guide vanes is employed to drive the moving vanes by expansion. The steam then passes to the next set of guide vanes and moving vanes and the process repeats itself until all the energy of the steam is realised. Fig 15 shows the arrangement of blades in Parsons' turbine

The Hamilton-Holzworth steam turbine resembles the Parsons in that the steam flows through the turbine in a continuous belt. But where as the steam in the Parsons type expands both in the guides vanes and in the wheel buckets, the expansion is confined to the guide vanes only in this type.

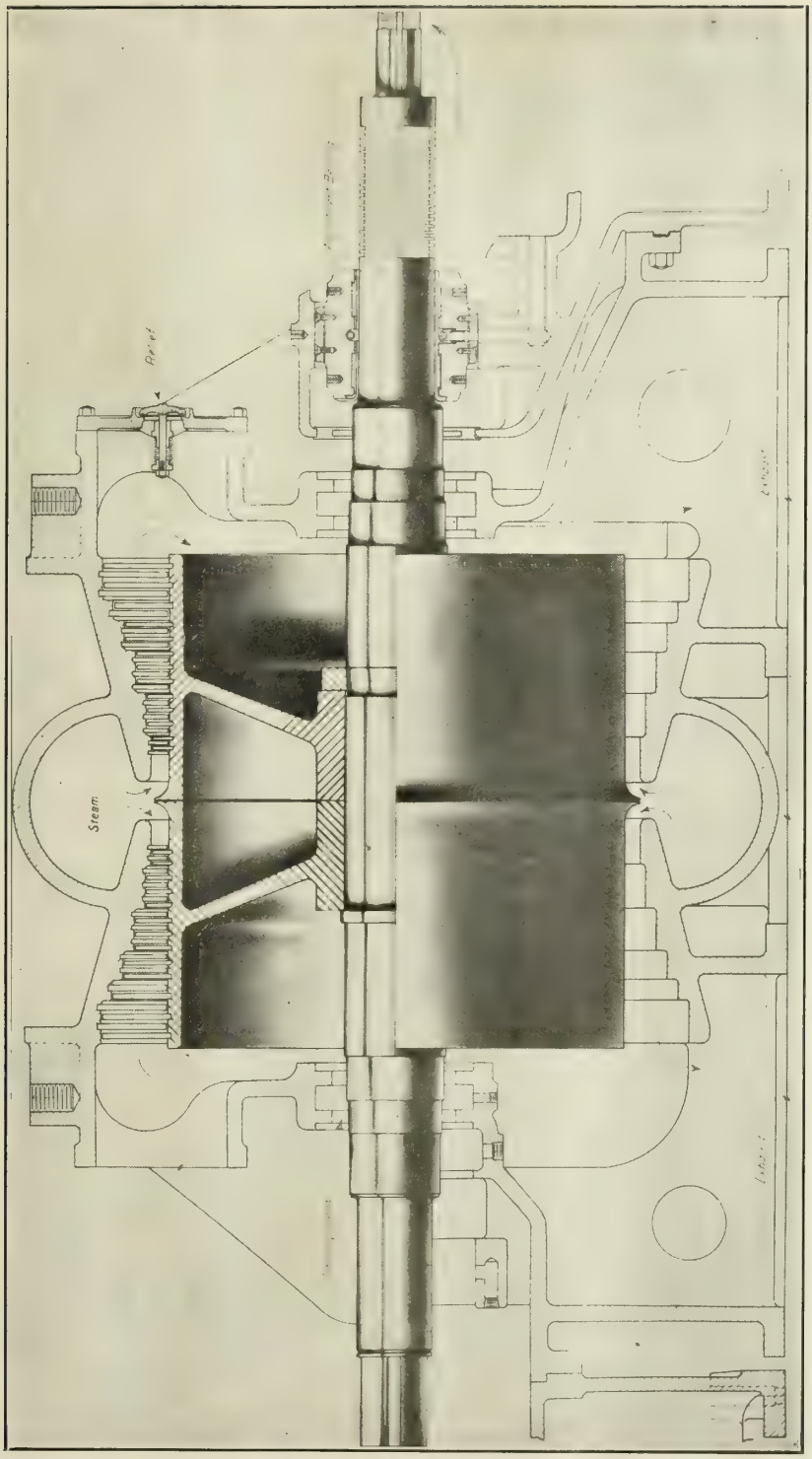
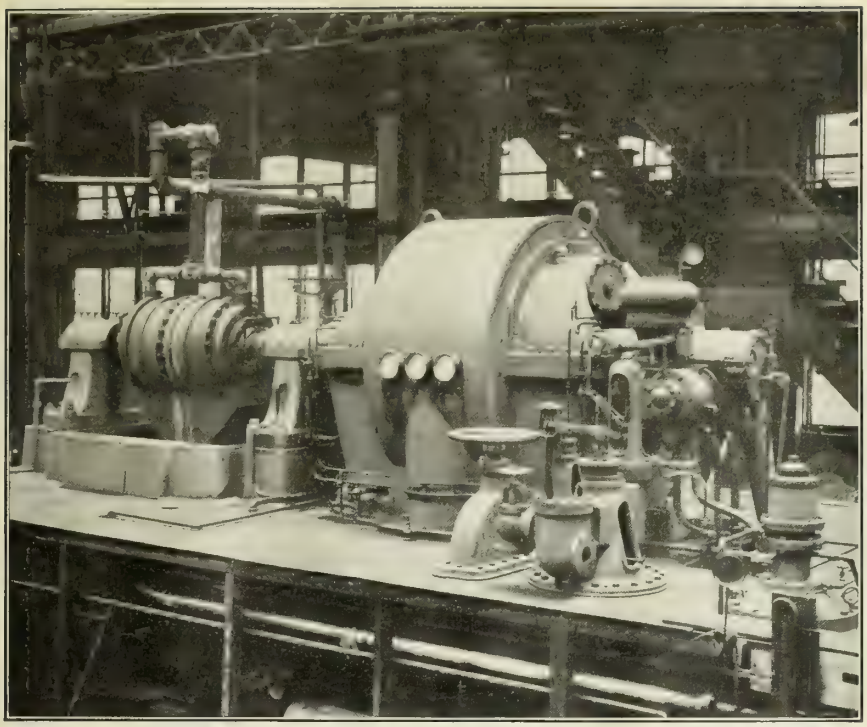


Fig. 16—Section, Westinghouse Low Pressure, Double-flow Turbine, 2,000 h.p., 15 lbs. absolute, 27.5 inches vacuum.



1,000 kw. Westinghouse, Double-flow, Low Pressure Turbine Unit on test floor; fitted with Governor and Automatic Reducing Valve. Arranged for brake load test.

Fig 17

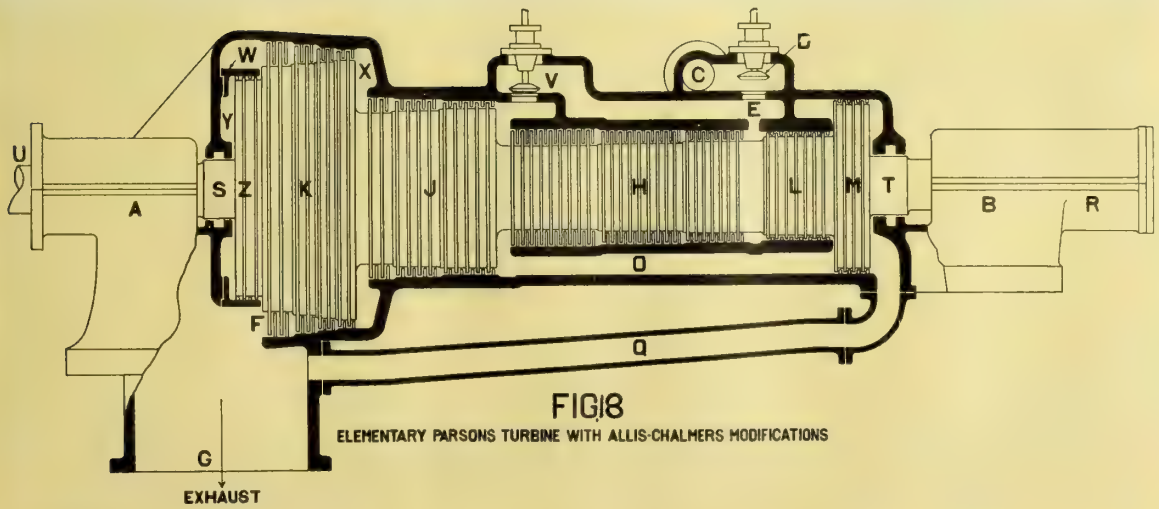


FIG 18

ELEMENTARY PARSONS TURBINE WITH ALLIS-CHALMERS MODIFICATIONS

Representative Low-pressure Turbine Installations

1. Gary Installation[3]: A typical example of low-pressure turbine application is found in the plant of of the United States Coal and Coke Co. Gary, west Va. Power is furnished for the mining property serving hoists, pumps, blowers &c. The plant contains two

Fig 19 Interior Power House, U.S. Coal & Coke Co., Gary, W. Va., showing Low Pressure Turbine.

24X44X22 Corliss Engines 750 K.W., one 100 K.W. Complete expansion steam turbine and one 1000 K.W. low-pressure steam turbine, both of Parsons type. These turbines are served by three cooling towers, fan driven, each 24 ft in diameter and 25 ft high. The following figures roughly indicate the normal operation of the units from readings taken october 7, 1908.

Output of engines - - - - - - - - - 1400 K.W.

Output of low-pressure turbine - 1200 K.W.

Steam pressure (gauge) - - - - - 150 pounds

Vacuum L.P. Turbine (refd to 30". 25.8 inches

Temperature of injection - - - - - 88 degrees

Temperature of air - - - - - - 85 degrees

Cooling in Tower - - - - - - - 31 degrees

From the above data it is seen that the low-pressure turbine carried nearly half the total load on less than 26 inches vacuum, and would have carried over 1500 K.W. on 28 inches vacuum.

2. Installation in plant of the Bettendorf Axle Company of Davenport, Ia.[4]: About two years ago this company was considering an addition to its power plant. After a thorough investigation of the various prime movers, all factors pointed to the low-pressure turbine as the most suitable power unit to install, from the stand-point of reliability.

Before the installation of the turbine the plant equipment was:

Two 100 K.W. direct connected high speed tandem compound units.

A number of hydraulic pumps.

An air compressor. All steam was exhausted into one header, making an ideal arrangement for

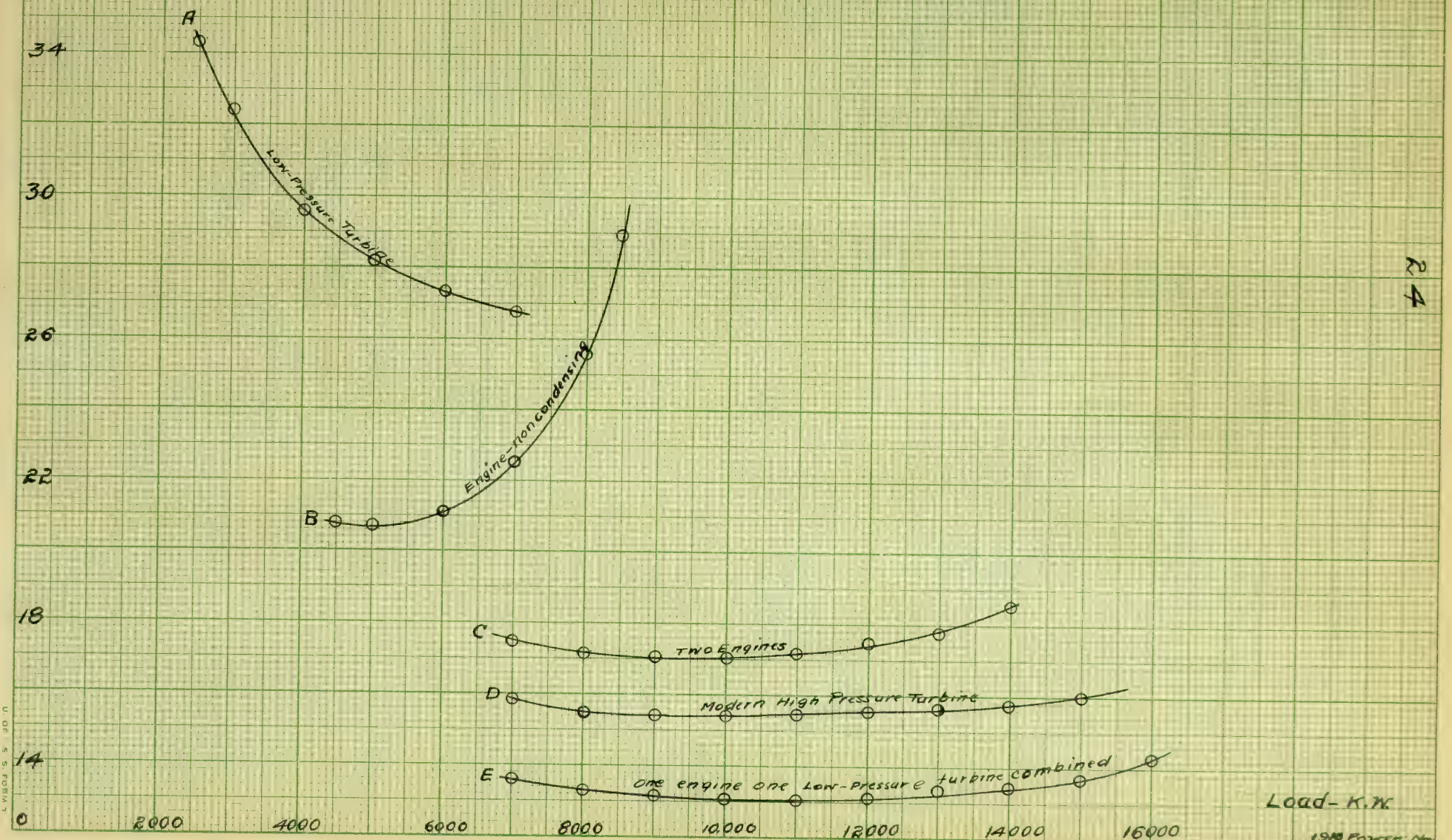
connection to an exhaust turbine.

The company decided to install a 500 K.W. horizontal Curtis turbine. The results of tests indicated that 75 to 80 percent of the energy delivered to the engines and auxiliaries was recovered. It was found that sufficient exhaust steam was available to furnish 425 K.W. continually as as much as 575 K.W. for short periods. The writer finally sums up "No shut-downs due to trouble of any kind with turbine, have occurred since the turbine was first put into operation, and this company can conscientiously recommend the installation of a low-pressure turbine in any place where a sufficient supply of exhaust steam is available for its operation."

3. The installation of Low-pressure turbine in the Interborough Rapid Transit Company in New York: This company has installed three 7500 K.W. units and is now engaged in installing two more units of the same size. Each of the 7500 K.W. low-pressure turbine units receives its steam from a Manhattan type of Corlis engine having two 42-inch horizontal high-pressure cylinders & two 86-inch vertical low-pressure cylinders. In Fig. 20, the curve A gives the low-pressure turbine water-rate, B gives engine water-rate running non-condensing, C

Pounds Dry Steam per K.W. Hr.

Fig 20



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gives the water-rate of two engines, E gives the water-rate of one engine and low-pressure turbine combined. D gives the wate-rate guaranteed for a typical modern 12000 K.W. standard high-pressure unit running under same conditions.

The results obtained by the installation of low-pressure turbines are as follows: (a) An increase of 100 percent in maximum capacity of plant (b) An increase of 146 percent in economic capacity of plant (c) A Saving of approximately 85 percent of the condensed steam for return to the boilers (d) An average improvement in economy of 13 percent over the best high-pressure turbine results (e) An average unit thrmal efficiency of 20.6 percent between the limits of 6500 and 15500 K.W.

Mr H G Stott concludes, "In general the installation of low-pressure turbines may be recommended wherever there are good engines already installed or in the case of a new installation where the load factor and the coal-cost are high. In plants having a large installation of a good type of reciprocating engine, the low-pressure turbine may be added at a total cost, including new condenser, auxitiaries, foundation, piping &c of not to exceed $25 per K.W., thus bringing down the average overall investment

per K.W. of the entire plant and so reducing the fixed charges per K.W.-hour."

4. Installation in the power house of Philadelphia Rapid Transit Company.: This station is part of the original Philadelphia Traction Company's power equipment and was built some years ago when the problem of high tension transmission was not solved. The station was equipped with direct current engine-generators and was located in the central part of the city with no water supply and consequently operated non-condensing. The load conditions being constant, its records compared favorably with stations operated condensing but under poorer load conditions

For a considerable period of time no thought was given to changing it to a condensing station as this involved numerous difficulties. After a time, however new stations with better equipment gave much better economy. At the same time the output of this station was increasing with no facilities to meet the demand. These two facts led to a consideration of the condensing problem. As no water was available it was necessary to install cooling towers. The installation of condensers was not practical as all available space had already been used. The engine-generators

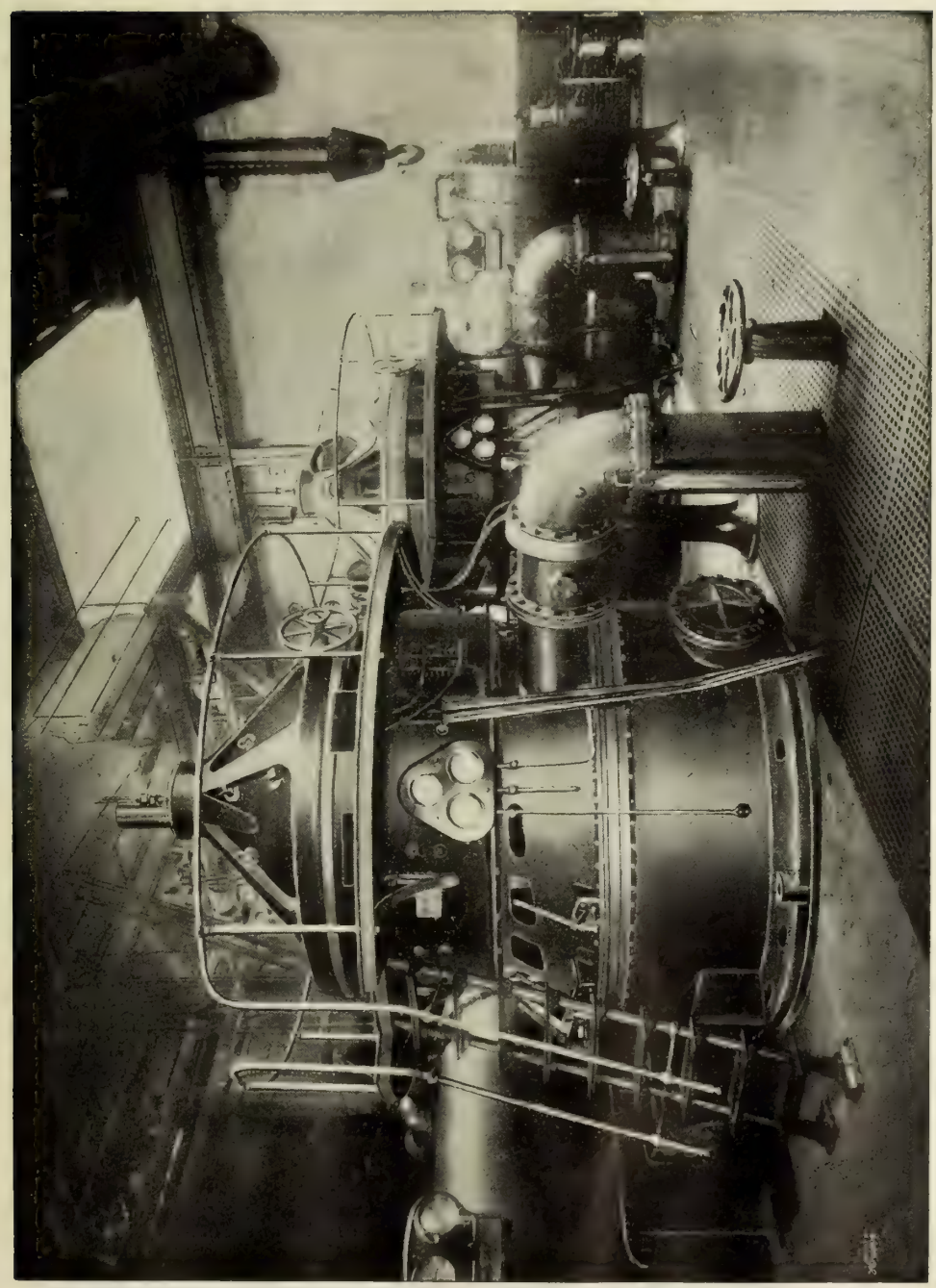

Fig. 21. LOW PRESSURE CURTIS TURBINE GENERATOR SETS (800-KW.) PHILADELPHIA RAPID TRANSIT CO.

were of insufficient size to stand any great increase of load for some time. Another difficulty was presented by the exhaust piping which had been in service a number of years and no provisions had ever been made to make it vacuum-tight. To run the engines condensing, the old exhaust pipes had to be replaced by new vacuum-tight pipes which meant considerable tearing out and additional expense.

About this time the exhaust steam turbine proposition presented itself. After considerable thought and figuring, it was finally decided to try an experimental installation at this plant. Two 800 K.W. direct current machines were purched and placed on the top of part of the sixth engine-foundation.

Original plant: The original design of the station provided for six Wetherill twin tandem Compound Corliss Engines 26×40×48 inches operating at 80 r.p.m. with 160 pounds initial steam pressure. Each pair of engines is direct connected to a 1500 K.W. D.C. generator. Four of these units were installed at first. The original heater equipment located in boiler room proved insufficient and a larger heater was needed. The only available room was in the engine room, and the heater was installed one one-half of the space allotted to the sixth unit. About this time

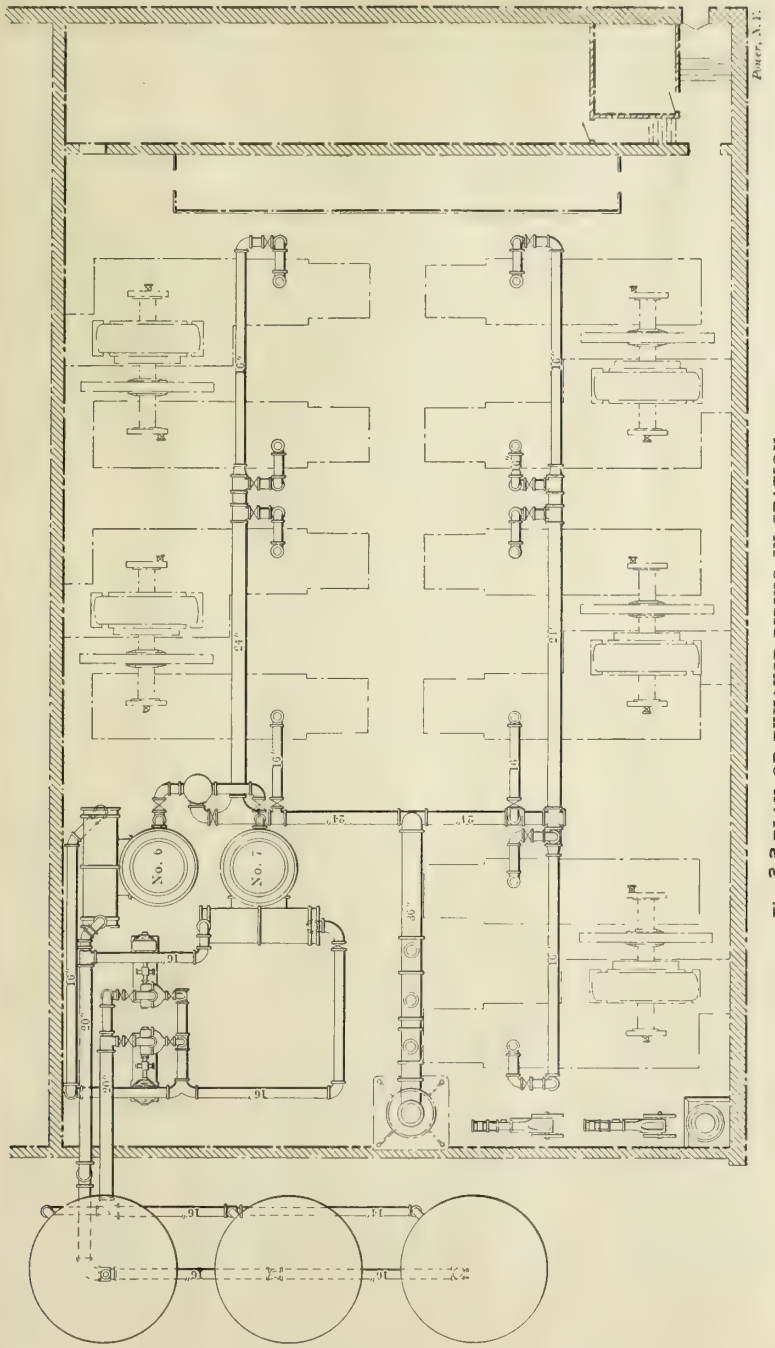

Fig. 22 PLAN OF EXHAUST PIPING IN STATION

the fifth unit was installed, which then completed the station as far as the original building was concerned it being impossible to place the sixth unit

The boiler room equipment had also reached its maximum at this time. This consisted of nineteen 375 Horse Power Babcox and Wilcox boilers and one 400 Horse Power Parker boiler

The main exhaust system consisted of two exhaust lines one on each side of the engine room basement, each designed to take care of three units

Details of Installation: In placing the exhaust steam turbines in this plant very few changes were made in the general scheme of exhaust piping. The only change made necessary by the installation of the turbines was to replace an ell by a tee in the 24 inch exhaust line on the east side of the station. The condensing equipment of the plant consists of two 8000 sq. ft counter-current Alberger surface condensers each connected directly to the turbine by means of a short make-up piece. Three cooling towers were erected against the north end of the building and carried on an elevated platform of reinforced concrete. Each tower is equipped with two 108 inch fans on opposite sides and operated by the same shaft driven by 40 Horse Power motors.

The turbines are of the Curtis vertical type and are of the same general design as the low-pressure end of the high-pressure Curtis turbine.

Operating Record: The first turbine went into operation on Dec 15 '1905, and the second one a little later. Fig 23 shows the record of these machines from

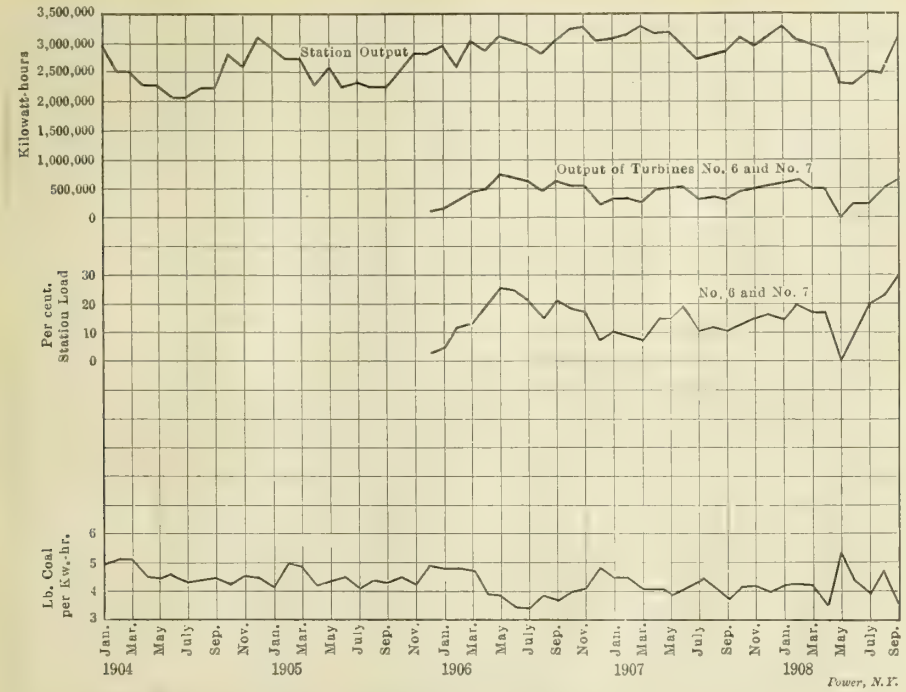

Fig. 23 TOTAL POWER OUTPUT AND PER CENT OF LOAD CARRIED BY EXHAUST STEAM TURBINES

that time up to September 8 '1908. The poor record during 1906-1907 is due to troubles with generators, principally with commutators. But even then, there was a considerable increase in the output and economy of the plant during that time. The average output of the machines during the years amounted to 14.6 percent of the engine load. By referring to the coal consumption curve, a marked decrease can be

seen during the period in which the turbines were in service The best records were always made when the turbines were in service the greater part of the time.

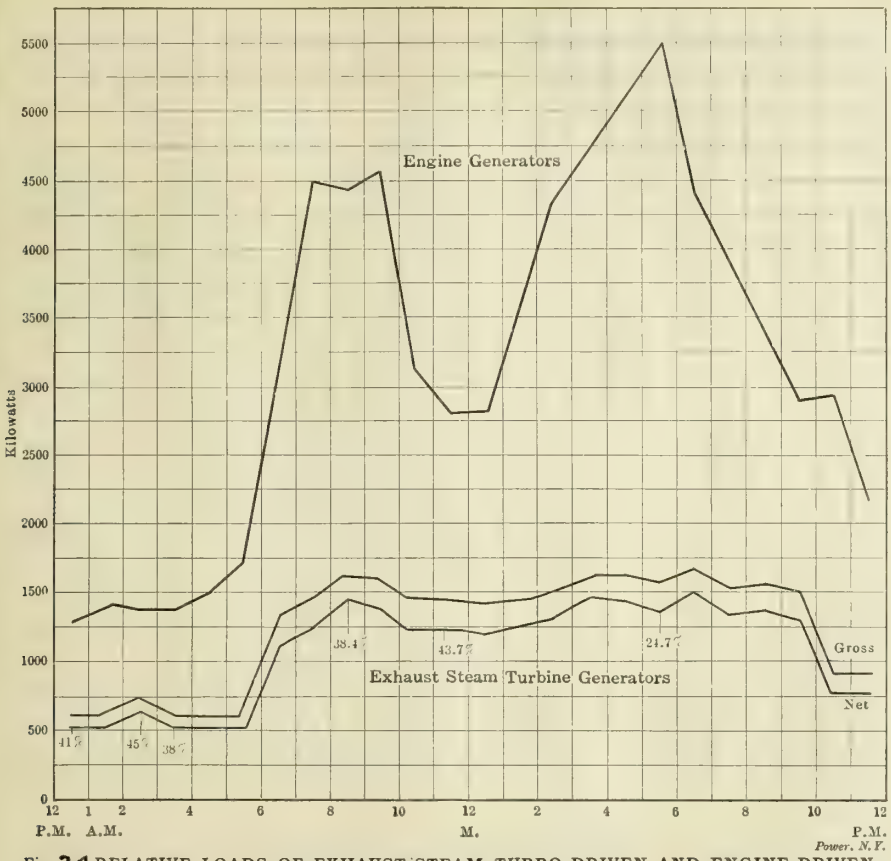

Fig. 24 RELATIVE LOADS OF EXHAUST STEAM TURBO-DRIVEN AND ENGINE-DRIVEN GENERATORS, SEPTEMBER 4, 1908

Fig 24 shows the out put curve for the station on September 4'1908 and is a typical curve during the Summer months. Fig 23 which shows the records of the turbines during 1906-1907 is some what misleading. While the turbines were operating under difficulties they still showed a gain of 14.6 prcent.

In order to show the reliability of the machines under present operating condition Fig. 25 has been

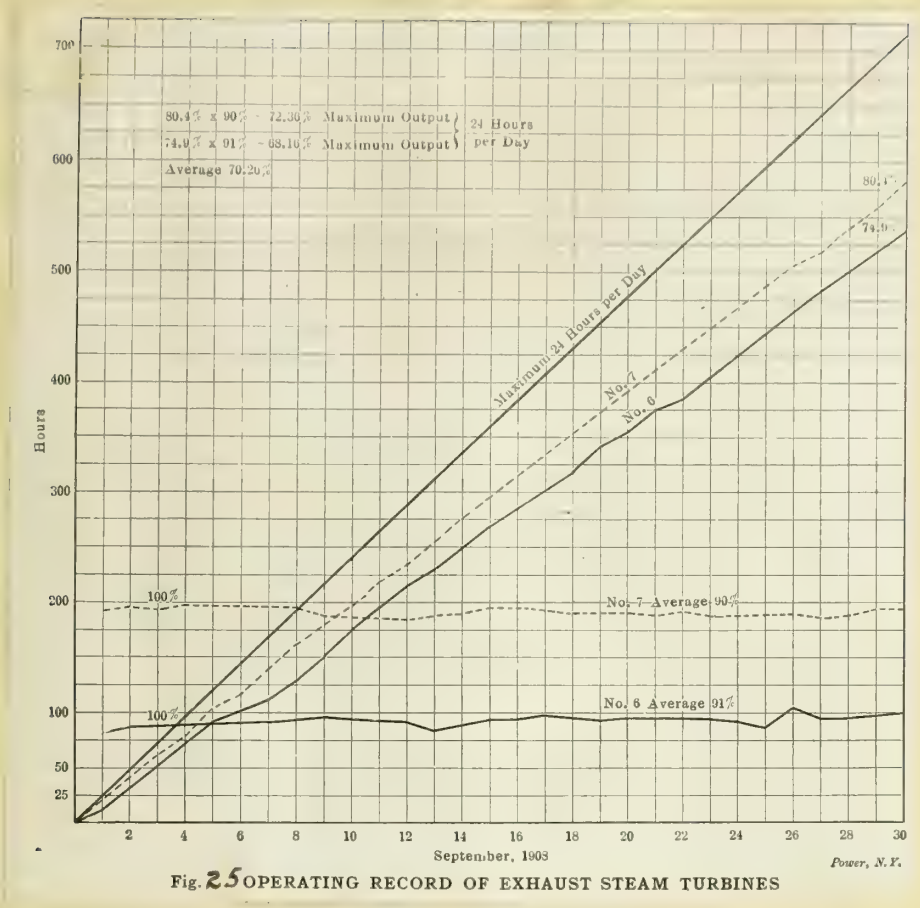

Fig. 25 OPERATING RECORD OF EXHAUST STEAM TURBINES

prepared. The curves show the record of the machines taken during September '1908. The practice at this station now is to keep the turbines in service the greatest period of time possible. The curve for No. 6 machine shows that it has been in service 74.9 percent of the entire time during the month and No. 7 machine 80.4 percent of the time. The two lower curves show the load factor, No. 6 averaging 91 percent full load and No. 7 90 percent full load for the entire month. By combining these load factors and the operating factors an average of 70.26 percent

for the two machines is obtained for the entire month of September.

In order to get some exact figures as to the performance of these machines tests were carried out and the results are shown in the following tables.

Table I Data on Turbine

Average load K.W.	805.85
Net steam used by turbine, per hr, lbs	30424
Steam per K.W.-hr (dry) lbs	37.75
Steam pressure pound abs	14.53
Vacuum, inches	27.50

Table II Auxiliaries

Power absorbed by motor driven aux. HP	158.02
Power absorbed by dry vacuum pump HP	9.94
Power absorbed by hot well pump HP	0.73
Power absorbed by step bearing pump H.P.	4.12
Total power absorved by auxiliaries H.P.	172.81
Percentage of auxiliaries to total output	15.9
Net increased load to engine load percent	16.88

Table III Condenser Results

Temperature of Air F°	78
Temperature of Injection water F°	87.6
Temperature of discharge water F°	98.5
Temperature of Hot well water F°	109.7

Assuming that the Corliss engines in this station take 25 lb of steam per I.h.p., it can be seen that if the entire exhaust steam was used one kilowatt could be developed at the turbine for each kilowatt out-put of the engine generators. Allowing 30 percent of exhaust for feed heaters and auxiliaries, the net output of the turbines approximate 70 percent of the engine capacity. The results obtained on the tests of this turbine are exactly in accordance with what was expected from regular station operating records and only confirms the belief that the exhaust steam turbine is worth considering as a means of increasing the economy and output of a non-condensing station

Taking the problem from a different point of view we find that the coal consumption of this station for the first six months of 1905 before the turbines were installed, averaged 4.48 pounds per kilowatt-hour. The coal consumption for first six months of 1906, after the turbines were installed averaged 4.0816 per kilowatt-hr, showing a saving of 0.4 lb of coal per kilowatt-hr. As the total output of the station for the first six months of 1906 was 20,346,890 kilowatt-hrs, this shows a saving of 4,069 tons of coal or 8138 tons for the year,

which figured at $3.00 per ton, amounts to $24 414.00.

As these two turbines are not using all the exhaust from the five engines, it is plain that when load requirements necessitate further increase, the installation of additional low-pressure turbines capable of utilising all the exhaust steam will cut the coal consumption down to 3 lb per kilowatt-hour.

From the point of view of first cost, the original steam equipment costs somewhere in the neighbourhood of $100 per kilowatt, and to have increased their output on its original lines would have required an investment directly proportional to the original investment. The low-pressure turbines, with cooling towers, however, was installed at an expense of approximately $50 per kilowatt. Thus a saving of 50 % in the future increase of plant is evident.

5. Combined Engine and Low-pressure turbine in the Merchant vessel "Otaki"

The "Otaki" is the first merchant vessel fitted with the combination system of reciprocating engines and turbines. The vessel was built for the New Zealand Shipping Company by Messrs Denny, of Dumbarton. She is virtually a sister ship to the twin-screw vessels "Orari" and "Opawa" fitted with reciprocating engines, which were built

by the same company. The boiler installations in the three vessels are identical; the working pressure is 200 pounds per sq. inch.

The engines of the "Otaki" consist of two sets of ordinary triple expansion reciprocating engines, driving wing propellers and a low-pressure turbine driving a center propeller. The turbine fitted in the "Otaki" is, generally speaking, of the ordinary Parsons' low-pressure type

Summary of Comparisons of Trials – Results of the "Otaki" and "Orari" for a speed of 14.6 knots.

Name	E.H.P.	I.H.P	Propulsive Coefficient	Water Consumption		
				Total per Hr.	Per E.H.P. per Hr.	Per I.H.P. per Hr.
Otaki	3350	5880	57 %	73330	21.9	13.7
Orari	3210	5360	60 %	88300	27.5	16.5
Gain percent in Otaki				17	20	17

The results show a decided gain in economy; but it is evident that an extensive adoption of this system will not be made, except where economy is the first consideration. The compactness and small floor space of turbines are great advantages and we see the great liners "Lusitania" and "Mauritania" equipped with turbines which are less economical than a combination of engines and low-pressure turbines.

6. Low-pressure turbine in the Liner "Olympic": The most remarkable application of low-pressure turbine is found on the White Star liner "Olympic" which is under construction. The arrangement of two wing propellers driven by reciprocating engines with a center propeller driven by a low-pressure turbine has been tried on the liner "Laurentic" with such gratifying results in economy and the elimination of vibration that it has been adapted for the liner "Olympic". The two wing propellers are driven by triple expansion engines with four cylinders the high pressure cylinder being 54 inches, the intermediate 84 inches and the two low-pressure cylinders 97 inches in diameter, while all have a stroke of 6 feet 3 inches. The center propeller is driven by the latest type of Parsons turbine taking steam from the low-pressure cylinders of the engines. From the company's standpoint the most attractive feature of this arrangement is that it keeps the coal consumption down which is always the most important item in marine practice except in fast boats where economy is sacrificed for high speed.

7. Power-plant of the Chicago North Western Railway Terminal. - First new plant which has installed low-pressure turbine in connection with engines:

In the engine room there are at present installed three vertical cross-compound non-condensing Allis-Chalmers Corliss Engines, with 25 and 44 by 42 inch cylinders, each rated at 1150 indicated horse power, at 100 r.p.m. with steam pressure at 155 lb and 60 degrees F. superheat. One Curtiss low-pressure steam turbine and generator unit of 500 k.w. capacity is installed in the machine room. It is the intention to run non-condensing during seasons when steam for heating will be necessary and use the low-pressure turbine only at such times as exhaust steam would ordinarily be allowed to waste to the almosphere. When not running on the heating system the light loads will be taken care of by one of the Corliss Engines and as the load increases beyond the capacity of one engine unit, it will be thrown on the low pressure turbine upto a point where the combined units are fully loaded, when another engine will be cut in and so on. The engines have a guaranteed water-rate of 18.6 pounds per I.H.P. against 16.5 pounds abo. back pressure. The turbine has a guaranteed water-rate of 32.7 pounds of exhaust steam per I.H.P. The engine at rated load would take (1150 x 18.6 = 21400) lb and develop 1150 I.H.P. The turbine

with same steam would develop (21400 ÷ 32.7 = 655) I.H.P. Therefore the total I.H.P. developed is equal to 1805 which gives a water-rate of 11.8 lb of steam per I.H.P.

Under present conditions it is possible to operate the turbine for eight months in the year and if the plans under consideration go through, the turbine will be used the year round. This is the first time an exhaust-steam turbine, installed in connection with engine units, has been included in the original layout.

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Heat Accumulators - Effect of Vacuum

In low-pressure steam turbine practice it frequently happens that the hourly supply of exhaust steam is equal to the hourly consumption of the turbine, but in many cases the supply is irregular and intermittent, owing to the fact that the high pressure machines may be hoisting engines in mines or rolling-mill engines, operating at short intervals of time

To steady the supply of exhaust steam and enable approximately constant steam to be obtained even if the supply of steam completely stops for short intervals of time, Professor Rateau has brought out a regenerative heat accumulator. This consists of a heat-storage reservoir in which the exhaust steam when in excess of the demand is condensed and in which the water is reevaporated when the exhaust steam supply fails or is insufficient. The two essentials of such an accumulator are that it shall contain sufficient mass of some material (generally water) to store the required amount of heat, and that there shall be a sufficient water surface for the necessary reevaporation without an undue fall in temperature. The accumulators work automatically when there is an excess of exhaust

steam; that is, when more steam is entering the accumulator than is being taken by the turbine the pressure rises. The boiling point, being dependent on the pressure, must rise also and consequently water can exist in the accumulator at a higher temperature. The water already in the accumulator forming the storage material absorbs heat from the steam which the latter gives up by, in part, condensing on the surface of the water, as due to the rise in the pressure of the steam, the water is a few degrees colder. The uncondensed steam passes to the low-pressure turbine.

When the steam supply stops or is insufficient, the pressure in the accumulator falls, owing to the constant demand of the turbine, and the boiling point falling in consequence, the water in the accumulator in part evaporates.[10]

These accumulators are, however suitable for smoothing out inequalities of steam supply of short duration. They are very suitable in cases of machines which run for a minute or part of a minute and stop for an equal period, but they are not economical in case of a stoppage or or serious diminution of steam supply of several minutes. This is due to the great size that is necessary

to store heat for anything more than short periods. But a large number of cases, come under the first class and heat accumulators may be used. In cases of long intervals mixed pressure turbines should be used instead of low-pressure turbines so that live steam may be used without any loss due to throttling.

The effect of vacuum is very evident from the theoretical watt-rate curves for different pressures, Fig. 26 The lower the condenser pressure, the lower would be the watt rate. This is especially true at the lower pressures as the curve indicates. But the most economical vacuum for a given plant involves a study of the plant economy at various vacua, the power consumption of auxiliaries, and the operating and fixed charges of the auxiliary plant.

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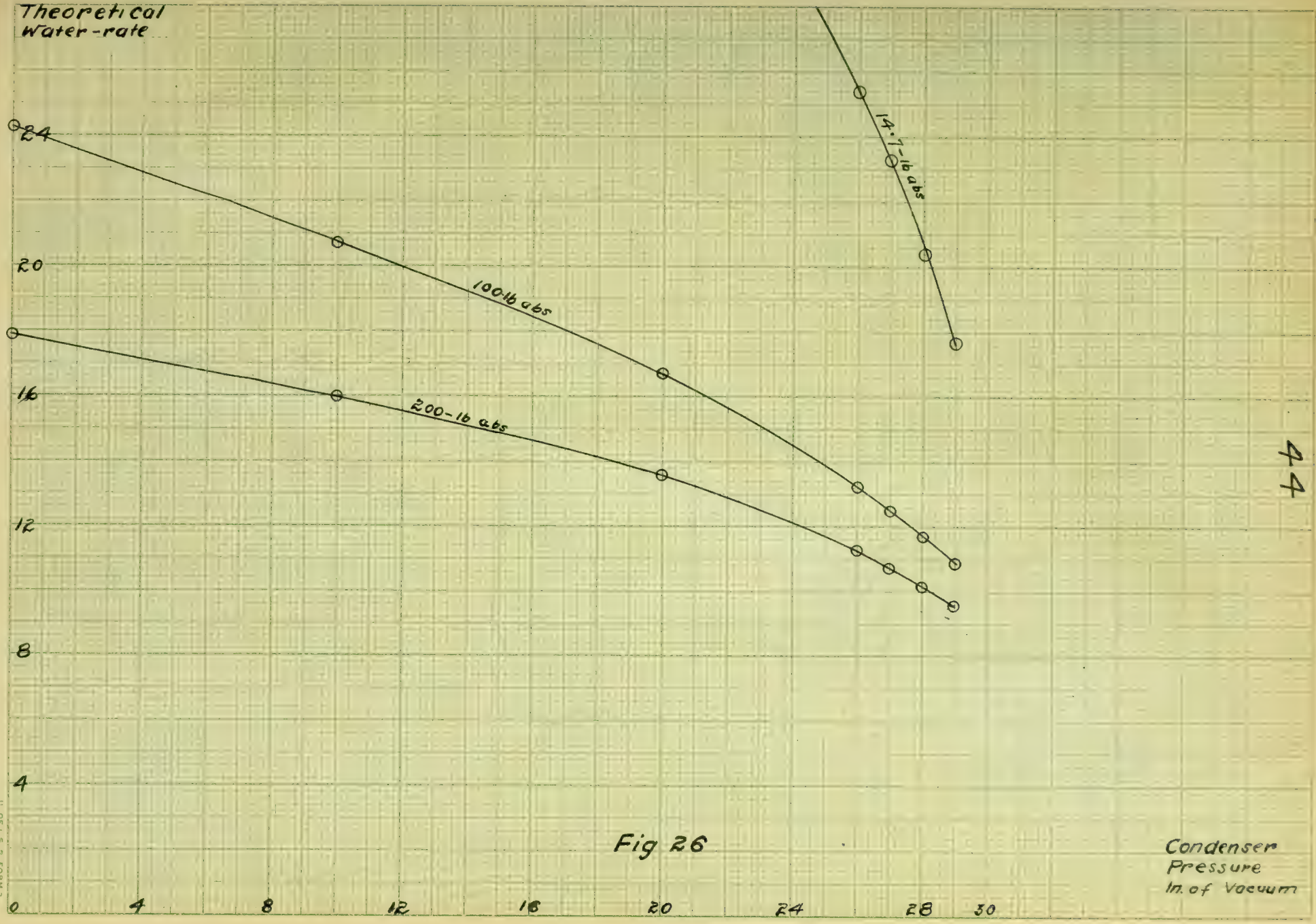

Fig 26

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Conclusion - Field and Future of Low-pressure Turbine

Taking up the different cases of low-pressure turbine installations and the economies derived from them in actual operation in various parts of this country and elsewhere, it seems very clear that the following are the possible applications:

1. Steam engines used intermittently on atmospheric exhaust.

2. Steam engines running continuously on atmospheric exhaust.

3. Steam engines in existing stations having ordinary condensing apparatus.

4. Marine propulsion of medium speed operation.

5. New stations in which the steam engine and the low-pressure turbine are designed for combined operation.

Of these five cases the first two seem to have the greatest field and thousands of engines are operating to-day non-condensing with only one-half of the total available work realised. In these cases an increase of 100 percent in the maximum capacity of the plant could be obtained by an additional first cost of one-third to one-half of its original investment per K.W. The additional power

is obtained from the exhaust steam without burning any more coal, consequently increasing the economy of the plant. The installation of the Rapid Transit Company in New York, which comes under the above cases resulted in an average thermal efficiency of 20.6 %, a remarkably high figure in steam practice

In the third case, although the saving is not so great as in the first two groups, yet in many instances of old design of plants it is economical to install low-pressure turbines and better condensing apparatus These facts are very applicable when such plants are crowded and there is no more room for additional engines and boilers. In the plant of the Philadelphia Rapid Transit Co. the installation of low-pressure turbines was the only solution of the problem of increasing plant output.

In marine propulsion it is very hard to say how the low-pressure turbine is going to be adopted. The compactness of the turbine and small floor space required for it, are sufficient advantages to replace all other kinds of prime movers. Yet it can be said with sufficient accuracy that in ordinary medium speed steamers, a combination of high pressure steam engine and low-pressure turbine gives

a decided economy and is worthy of consideration. The recent installations of this type of machinery in the big liners "Laurentic" and "Olympic" go to prove the same fact.

In new stations it is extremely doubtful whether the combination of high pressure engines and low-pressure turbine is better than steam-turbine alone. The writer has been able to find only one case where low-pressure turbine has been installed in connection with high pressure engines. In the power plant of Chicago North Western Railway Terminal, the calculation shows good economy and gives a water-rate of 11.8 lbs per I.H.P. In new stations, however, the simplicity afforded by one type of machine is considered an advantage and overshadows other considerations.

But whatever has been said of the low-pressure turbine, it seems at first that it is paradoxical, getting energy out of nothing and disproving the fundamental law of thermodynamics. Practically it amounts to the same thing, the energy which was formerly wasted in the exhaust steam, is harnessed in the low-pressure turbine and an equal amount of work is obtained. There is no doubt, however, that the low-pressure turbine is the most efficient

prime-mover of all. The high pressure turbine cannot give good economy on account of the friction of blades and the reciprocating engine which is highly efficient at high pressure cannot get the advantage of the last stages of expansion on account of the large specific volume of steam. It is not only a good machine by itself, but when it combines with the reciprocating engine it helps to get power out where engine is helpless. The high-pressure steam turbine would have been a practical failure and the reciprocating engine would have been the best prime-mover to-day if the low-pressure end of the turbine were cut out.

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